

A new combined cooling, heating and power system driven by solar energy

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ABSTRACT

A new combined cooling, heating and power (CCHP) system is proposed. This system is driven by solar energy, which is different from the current CCHP systems with gas turbine or engine as prime movers. This system combines a Rankine cycle and an ejector refrigeration cycle, which could produce cooling output, heating output and power output simultaneously. The effects of hour angle and the slope angle of the aperture plane for the solar collectors on the system performance are examined. Parametric optimization is conducted by means of genetic algorithm (GA) to find the maximum exergy efficiency. It is shown that the optimal slope angle of the aperture plane for the solar collectors is 60° at 10 a.m. on June 12, and the CCHP system can reach its optimal performance with the slope angle of 45° for the aperture plane at midday. It is also shown that the system can reach the maximum exergy efficiency of 60.33% under the conditions of the optimal slope angle and hour angle.

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1. Introduction

The trigeneration concept refers to the possibility of efficiently and profitably combining cooling, heating and power production (CCHP) to satisfy the consumption needs during the energy system operation. In a competitive energy market framework, the adoption of CCHP systems may become profitable with respect to traditional systems, where electricity, heat, and cooling are produced or purchased separately.

There are several current CCHP technologies to perform the trigeneration according to the prime movers, such as steam turbine, reciprocating internal combustion engine, gas turbine, micro gas turbine, stirling engine, fuel cell. Wang et al. [1] had made an extensive and an intensive review of CCHP in the literature. Wang et al. [2,3] also investigated the CCHP system driven by stirling engine and gas turbine. Li et al. [4] conducted the energy utilization evaluation for the CCHP systems mainly with gas engine and gas turbine as prime movers.

Although steam turbine, reciprocating internal combustion engine and gas turbine that can be considered as the conventional prime movers still make up most of the gross capacity being installed, and micro gas turbine, Stirling engine and fuel cell present a promising future for prime movers in CCHP system, the current CCHP

technologies must inevitably consume the fossil fuels, which would disappear with energy consumption and economic development in the future. So, it is necessary to utilize renewable energy.

The main objective of the present study is to propose a new combined cooling, heating and power (CCHP) system driven by solar energy. This system combines Rankine cycle and ejector refrigeration cycle, and produces cooling output, heating output and power output simultaneously. The ejector refrigeration cycle, which many studies [5–11] have been devoted to, is different from the absorption refrigeration cycle in most of CCHP systems. It has some advantages such as fewer movable parts and low operating, installation and maintenance cost. In addition, the ejector refrigeration cycle has the possibility of using a wide range of refrigerants with the system.

In the present study, the simulation of the CCHP system driven by solar energy is achieved using a simulation program. The effects of hour angle and the slope angle of the aperture plane for the solar collectors on the system performance are examined. In addition, the parameter optimization is achieved with exergy efficiency as the objective function by means of genetic algorithm under the given condition.

2. System description

The proposed CCHP system is driven by solar energy, and combines the Rankine cycle and the ejector refrigeration cycle, which could produce cooling, heating and power simultaneously. Fig. 1 illustrates the CCHP system. The overall system is divided into

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Nomenclature

A	surface area of solar collectors, m^2
b	width of absorber, m
c	heat capacity, kJ kg^{-1}
C	concentration ratio
D	diameter, m
E	exergy, kW
h	enthalpy, kJ kg^{-1}
H	height, m
I	hourly radiation, W m^{-2}
k	heat transferring coefficient, $\text{W m}^{-2} \text{K}^{-1}$
L	length of tube, m
m	mass flow rate, kg s^{-1}
ma	average number of reflections
n	the day of the year
N	number of tubes
p	pressure, MPa
Q	heat load, kW
R	tilt factor; ratio
s	entropy, $\text{kJ kg}^{-1} \text{K}^{-1}$
S	total effective flux, W m^{-2}
t	temperature, $^{\circ}\text{C}$; time
T	temperature, K
u	velocity, m s^{-1}
U	overall loss coefficient, $\text{W m}^{-2} \text{K}^{-1}$
W	width, m

Greek letters

α	absorptivity of the absorber surface
β	slope angle of the aperture plane for solar collectors, $^{\circ}$
δ	declination
η	efficiency
θ	angle of incidence
μ	entrainment ratio
ρ	reflectivity of the concentrator surface; density, kg m^{-3}
τ	transmissivity of the cover
ω	hour angle, $^{\circ}$
φ	latitude, $^{\circ}$

Subscripts

a	acceptance half angle
b	beam
br	branching
B	boiler
C	condenser
d	diffuse radiation; diffuser section of ejector
e	effective
exg	exergy
extr	extraction
E	evaporator
fi	inlet of solar collectors
fo	outlet of solar collectors
g	global
hc	average temperature of heater
H	heater
i	Inlet
inst	Instantaneous
l	uniform
lo	Loss
m	mixing section of ejector
mf	mixed fluid
n	nozzle
n1	inlet of nozzle
n2	outlet of nozzle
NET	Net
p	pressure
pf	primary flow
P1	pump 1
P2	pump 2
s	isentropic process
sc	average temperature of solar collectors
sf	secondary flow
thm	thermal
T	turbine
u	useful
z	zenith angle
0	environment

two subsystems: the solar collector subsystem and the CCHP subsystem.

2.1. The solar collector subsystem

This subsystem consists of solar collectors, a thermal storage tank and an auxiliary heater. The solar collectors are used as a main energy source to supply heat to the system. The thermal storage tank is used as the thermal *source* when solar radiation is not sufficient. The auxiliary heater is installed as the backup energy source to boost the temperature of thermal storage tank to the allowable reference temperature when the temperature of thermal storage tank drops below the allowable reference temperature.

A compound parabolic collector (CPC) is used to collect the solar radiation because it achieves higher concentration for large acceptance angle and requires only intermittent sun-tracking. In addition, the CPC can achieve the higher temperature than flat plate collector for power generation. In the present study, the symmetric CPC with flat absorber is placed on a horizontal east–west axis and oriented with its aperture plane sloping at an angle of 45° .

The hourly global radiation I_g reaching a horizontal surface on the earth is given by:

$$I_g = I_b + I_d \quad (1)$$

where

$$I_b = I_{bn} \cos \theta_z \quad (2)$$

where I_{bn} is beam radiation in the direction of the rays, and θ_z is angle of incidence on a horizontal surface, i.e. the zenith angle.

It is postulated that for a clear cloudless day:

$$I_{bn} = A \exp \left[-\frac{B}{\cos \theta_z} \right] \quad (3)$$

and

$$I_d = C I_{bn} \quad (4)$$

where A , B and C are constants [12].

Since CPC for absorbing radiation is tilted at an angle to the horizontal, it is necessary to calculate the tilt factor for radiation.

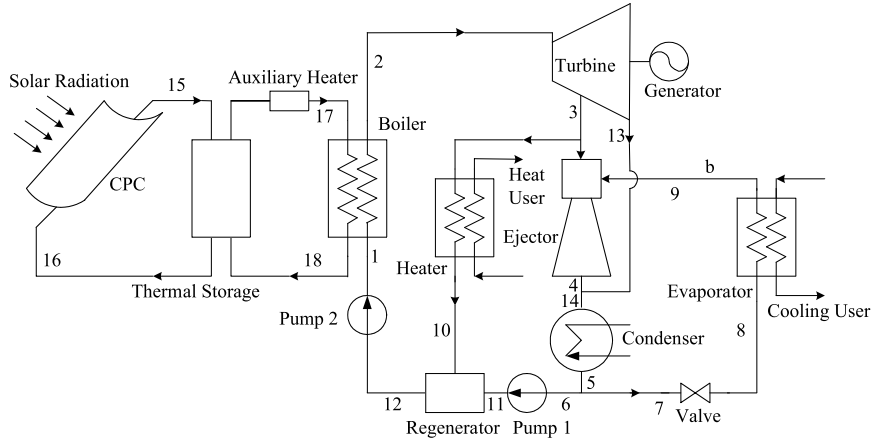


Fig. 1. Schematic diagram of CCHP system driven by solar energy.

The ratio of the beam radiation flux falling on a tilted surface to that falling on a horizontal surface is called the tilt factor R_b for beam radiation, which can be given by:

$$R_b = \frac{\cos \theta}{\cos \theta_z} = \frac{\sin \delta \sin(\phi - \beta) + \cos \delta \cos \omega \cos(\phi - \beta)}{\sin \delta \sin \phi + \cos \delta \cos \omega \cos \phi} \quad (5)$$

where the declination δ can be given by Cooper [13].

$$\delta = 23.45 \sin \left[\frac{360}{365} (284 + n) \right] \quad (6)$$

The tilt factor R_d for diffuse radiation is the ratio of the diffuse radiation flux falling on the tilted surface to that falling on a horizontal surface. The value of this tilt factor depends upon the sky dome seen by the tilted surface. Assuming that the sky is an isotropic source of diffuse radiation, we have

$$R_d = \frac{1 + \cos \beta}{2} \quad (7)$$

since $(1 + \cos \beta)/2$ is the radiation shape factor for a tilted surface with respect to the sky.

The geometry of a two-dimensional CPC is shown in Fig. 2. The concentration ratio is given by:

$$C = \frac{W}{b} = \frac{1}{\sin \theta_a} \quad (8)$$

where θ_a is the acceptance half angle.

The height to aperture ratio of the concentrator is given by:

$$\frac{H}{W} = \frac{1}{2} \left(1 + \frac{1}{\sin \theta_a} \right) \cos \theta_a \quad (9)$$

Rabl [14] has shown that the ratio of the surface area of the concentrator to the area of the aperture is given by the expression:

$$\frac{A}{WL} = \sin \theta_a (1 + \sin \theta_a) \left[\frac{\cos \theta_a}{\sin^2 \theta_a} + \ln \left\{ \frac{(1 + \sin \theta_a)(1 + \cos \theta_a)}{\sin \theta_a [\cos \theta_a + (2 + 2 \sin \theta_a)^{0.5}]} \right\} - \frac{\sqrt{2} \cos \theta_a}{(1 + \sin \theta_a)^{1.5}} \right] \quad (10)$$

Rabl has also shown that the average number of reflections ma undergone by all radiation falling within the acceptance angle before reaching the absorber surface is given by the expression

$$ma = \frac{1}{2 \sin \theta_a} \left(\frac{A}{WL} \right) - \frac{(1 - \sin \theta_a)(1 + 2 \sin \theta_a)}{2 \sin^2 \theta_a} \quad (11)$$

Thus, the effective reflectivity of the concentrator surface is given by:

$$\rho_e = \rho^{ma} \quad (12)$$

where ρ_e is the effective reflectivity of the concentrator surface for all radiation, and ρ is the reflectivity value for a single reflection.

Because of its large acceptance angle, a CPC accepts both beam and diffuse radiation. The beam radiation flux falling on the aperture plane is $I_b R_b$, while the diffuse radiation flux within the acceptance angle is given by (I_d/C) . Thus, the total effective flux absorbed at the absorber surface is:

$$S = \left[I_b R_b + \frac{I_d}{C} \right] \tau \rho_e \alpha \quad (13)$$

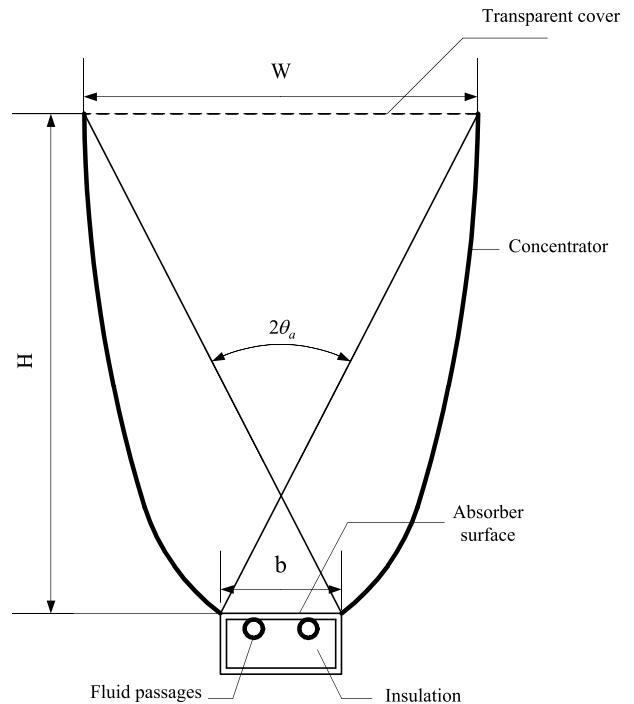


Fig. 2. Compound parabolic collector.

The useful heat gain rate can be given by:

$$Q_u = F_R WL \left[S - \frac{U_{lo}}{C} (T_{fi} - T_0) \right] \quad (14)$$

where

$$F_R = \frac{mc_p}{bU_{lo}L} \left\{ 1 - \exp \left[\frac{F' b U_{lo} L}{mc_p} \right] \right\} \quad (15)$$

and

$$\frac{1}{F'} = U_{lo} \left[\frac{1}{U_{lo}} + \frac{b}{N\pi D_i k} \right] \quad (16)$$

The instantaneous collection efficiency is given by:

$$\eta_{inst} = \frac{Q_u}{(I_b R_b + I_d R_d) WL} \quad (17)$$

The insulated liquid storage tank with sensible heat storage is shown in Fig. 3, receiving energy from an array of collectors and discharging energy to a load for use in the CCHP subsystem. We assume that the liquid in the tank is always well-mixed and consequently is at T_i . T_i is defined as the water temperature in the tank after two streams from solar collector and boiler are well-mixed. An energy balance on the thermal storage tank yields the following equation [12]:

$$[(\rho V C_p)_l + (\rho V C_p)_t] \frac{dT_i}{dt} = Q_u - Q_{load} - (UA)_t (T_i - T_0) \quad (18)$$

where $(\rho V C_p)_l$ represents the heat capacity of the liquid in the tank, $(\rho V C_p)_t$, the heat capacity of the tank material, Q_u , the rate of useful heat gain received from the collectors, Q_{load} , the rate at which energy is being discharged to the load, $(UA)_t$, the product of the overall heat-transfer coefficient and surface area of the tank. The heat capacity term $(\rho V C_p)_t$ is likely to be of importance for small-sized tanks only. For large tanks, its value may be negligible in comparison to $(\rho V C_p)_l$. Denoting the sum of the two heat capacities by the symbol $(\rho V C_p)_e$ and integrating differential Eq. (18) under the assumption that Q_u , Q_{load} and T_0 are constants, we get, subject to the initial condition $t = 0$, $T_i = T_i$ [12]:

$$\frac{Q_u - Q_{load} - (UA)_t (T_i - T_0)}{Q_u - Q_{load} - (UA)_t (T_i - T_0)} = \exp \left[- \frac{(UA)_t t}{(\rho V C_p)_e} \right] \quad (19)$$

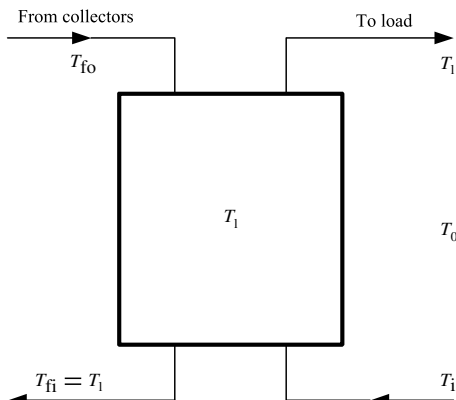


Fig. 3. A well-mixed sensible heat liquid storage tank.

The values of Q_{load} may be calculated from the inlet and outlet temperature and flow rate as:

$$Q_{load} = m_{load} c_p (T_l - T_i) \quad (20)$$

2.2. The CCHP subsystem

The CCHP subsystem consists of a few major components: boiler, turbine, evaporator, heater, regenerator, condenser, ejector, pumps and throttle valve. The boiler is a device in which high pressure and temperature vapor is generated by absorbing heat from thermal storage system. The high pressure and temperature vapor are expanded through the turbine to generate power. The extracted vapor from the turbine is divided into two streams. One stream enters the heater to supply heat load to heat user. The other stream as the primary vapor enters the supersonic nozzle of ejector. The very high velocity vapor at the exit of the nozzle produces a high vacuum at the inlet of the mixing chamber and entrains secondary vapor into the chamber from the evaporator. The two streams are mixed in the mixing chamber. Then the mixed stream becomes a transient supersonic stream. On entering the constant cross-section zone, a normal shock wave occurs, accompanied by a significant pressure rise. After the shock, the velocity of the mixed stream becomes subsonic and decelerates in the diffuser. The stream from ejector and the turbine exhaust are mixed together, and they enter the condenser where they condense from vapor to a liquid by rejecting heat to the surroundings. One part of the working fluid leaving the condenser enters the evaporator after passing through the throttle valve, and the other part flows to the regenerator after passing through one pump. The stream with relatively high temperature leaving the heater heats the stream from a pump in the regenerator where the two streams are mixed together. The pressure of the liquid leaving the regenerator is increased to the boiler pressure by the other pump, and it is pumped to the boiler to be vaporized again. The working fluid entering the evaporator is at low pressure and temperature. In the evaporator, it is vaporized by absorbing heat from the cooled media. Thus, a cooling effect is produced.

The ejector is the key component in this CCHP system. In the present study, the ejector performance simulation is carried out based on the one-dimensional constant pressure flow model used by most researchers in studying ejector refrigeration systems. The basic principle of the model was introduced by Keenan et al. [15] based on gas dynamics, and developed by Munday et al. [16], Huang et al. [17] and Ouzzane et al. [18]. The following assumptions are made for the analysis.

- (1) The flow inside the ejector is in steady state and one-dimensional.
- (2) The primary and secondary flows at the inlet of the ejector and the mixed flow at the outlet of the ejector are at stagnation conditions.
- (3) For simplicity, the effects of frictional and mixing losses in the nozzle, diffuser and mixing section are taken into account by using the nozzle efficiency, the mixing efficiency and the diffuser efficiency.
- (4) Whenever the mixed flow is supersonic, a normal shock wave is assumed to occur upstream of diffuser inlet.
- (5) Mixing process in the mixing section of ejector occurs at constant pressure and complies with the conservation of energy and momentum.
- (6) The ejector does not exchange heat with the surroundings.

Based on the above assumptions, the mass, momentum and energy equations are applied to sections for the ejector and then the following equations are obtained.

In the nozzle section, the energy conservation equation for the adiabatic and steady primary flow is given as:

$$h_{pf,n2} + u_{pf,n2}^2/2 = h_{pf,n1} + u_{pf,n1}^2/2 \quad (21)$$

Compared with the outlet velocity of primary flow $u_{pf,n2}$, the inlet velocity of primary flow $u_{pf,n1}$ could be negligible. According to Eq. (21), the outlet velocity of primary flow $u_{pf,n2}$ is expressed as:

$$u_{pf,n2} = \sqrt{2(h_{pf,n1} - h_{pf,n2})} \quad (22)$$

The nozzle efficiency is given as:

$$\eta_n = \frac{h_{pf,n1} - h_{pf,n2}}{h_{pf,n1} - h_{pf,n2,s}} \quad (23)$$

hence

$$u_{pf,n2} = \sqrt{2\eta_n(h_{pf,n1} - h_{pf,n2,s})} \quad (24)$$

The entrainment ratio of ejector is given as:

$$\mu = \frac{m_{sf}}{m_{pf}} \quad (25)$$

In the mixing section, the momentum conservation equation is:

$$m_{pf}u_{pf,n2} + m_{sf}u_{sf,n2} = (m_{pf} + m_{sf})u_{mf,m,s} \quad (26)$$

hence

$$u_{mf,m,s} = \frac{u_{pf,n2} + \mu \cdot u_{sf,n2}}{1 + \mu} \quad (27)$$

where

$$u_{sf,n2} = \sqrt{2(h_{sf} - h_{sf,n2})} \quad (28)$$

Compared with the primary flow velocity $u_{pf,n2}$, the velocity of secondary flow $u_{sf,n2}$ is negligible. Then Eq. (27) becomes:

$$u_{mf,m,s} = \frac{u_{pf,n2}}{1 + \mu} \quad (29)$$

The mixing efficiency is given as:

$$\eta_m = \frac{u_{mf,m}^2}{u_{mf,m,s}^2} \quad (30)$$

The average velocity of mixed flow is written according to Eqs. (29) and (30) as:

$$u_{mf,m} = u_{pf,n2} \sqrt{\eta_m} / (1 + \mu) \quad (31)$$

The energy conservation equation for the mixing section is:

$$\begin{aligned} m_{pf} \left(h_{pf,n2} + \frac{u_{pf,n2}^2}{2} \right) + m_{sf} \left(h_{sf,n2} + \frac{u_{sf,n2}^2}{2} \right) \\ = (m_{pf} + m_{sf}) \left(h_{mf,m} + \frac{u_{mf,m}^2}{2} \right) \end{aligned} \quad (32)$$

The enthalpy of mixed flow is written according to Eqs. (22), (28) and (32) as:

$$h_{mf,m} = (h_{pf,n1} + \mu h_{sf}) / (1 + \mu) - u_{mf,m}^2/2 \quad (33)$$

In diffuser section, the mixed fluid converts the kinetic energy into pressure energy. Assuming the compression process is isentropic, the energy equation for the diffuser section is:

$$\frac{1}{2}(u_{mf,m}^2 - u_{mf,d,s}^2) = h_{mf,d,s} - h_{mf,m} \quad (34)$$

The actual energy equation for the diffuser section is:

$$\frac{1}{2}(u_{mf,m}^2 - u_{mf,d}^2) = h_{mf,d} - h_{mf,m} \quad (35)$$

The diffuser efficiency is given as:

$$\eta_d = \frac{h_{mf,d,s} - h_{mf,m}}{h_{mf,d} - h_{mf,m}} \quad (36)$$

Compared with the velocity $u_{mf,m}$, the outlet velocity of the mixed fluid $u_{mf,d}$ can be neglected, the actual outlet enthalpy of the mixed fluid $h_{mf,d}$ is written according to energy Eq. (35) as:

$$h_{mf,d} = h_{mf,m} + u_{mf,m}^2/2 \quad (37)$$

In addition, according to Eq. (36), the actual outlet enthalpy of the mixed fluid is also expressed as:

$$h_{mf,d} = h_{mf,m} + (h_{mf,d,s} - h_{mf,m}) / \eta_d \quad (38)$$

It is known that the performance of ejector is evaluated by its entrainment ratio μ . Based on above Eqs. (24), (31) and (38), it can be derived as:

$$\mu = \sqrt{\eta_n \eta_m \eta_d (h_{pf,n1} - h_{pf,n2,s}) / (h_{mf,d,s} - h_{mf,m})} - 1 \quad (39)$$

When the inlet state parameters of primary flow, secondary flow and back pressure of the ejector are given, the value of entrainment ratio μ could be found using iterative calculation.

Extraction mass flow ratio is defined as the extraction mass flow divided by the turbine inlet mass flow, given by:

$$R_{extr} = \frac{m_{extr}}{m_T} \quad (40)$$

Mass flow branching ratio is defined as the primary mass flow of ejector divided by the extraction mass flow, given by:

$$R_{br} = \frac{m_{pf}}{m_{extr}} \quad (41)$$

The basic equations obtained from the conservation law for energy in the components are written as follows:

$$\text{For evaporator : } Q_E = m_E(h_9 - h_8) \quad (42)$$

$$\text{For boiler : } Q_B = m_B(h_2 - h_1) \quad (43)$$

$$\text{For condenser : } Q_C = m_C(h_{14} - h_5) \quad (44)$$

$$\text{For turbine : } W_T = m_T(h_2 - h_3) + (m_T - m_{extr})(h_3 - h_{13}) \quad (45)$$

$$\text{For pump 1: } W_{P1} = m_{P1}(h_{11} - h_6) \quad (46)$$

$$\text{For pump 2: } W_{P2} = m_{P2}(h_1 - h_{12}) \quad (47)$$

$$\text{For heater: } Q_H = m_H(h_3 - h_{10}) \quad (48)$$

$$\text{For regenerator: } m_{P2}h_{12} = m_Hh_{10} + m_{P1}h_{11} \quad (49)$$

2.3. System performance

The overall system performance can be evaluated by the thermal efficiency and exergy efficiency. Thermal efficiency of the overall system is defined as the useful energy output divided by the total solar energy input, given by:

$$\eta_{\text{thm}} = \frac{W_{\text{NET}} + Q_E + Q_H}{Q_u} \quad (50)$$

where W_{NET} is the power output from the turbine, reduced by the power input to the pump, Q_E is the cooling output, Q_H is the heat output and Q_u is the total heat added to the system from the solar collectors.

From the viewpoint of the first law of thermodynamics and energy conservation used to determine the overall thermal efficiency, work and heat are equivalent. On the other hand, exergy, based on the second law of thermodynamics, quantifies the difference among work, heat and cooling in terms of irreversibility, or change in energy quality. Therefore, the exergy efficiency is chosen to be the criterion for the overall system performance evaluation.

Exergy efficiency is defined as the exergy output divided by the exergy input to the overall system, which can be given by:

$$\eta_{\text{exg}} = \frac{W_{\text{NET}} + E_E + E_H}{E_u} \quad (51)$$

E_u is the exergy input to the overall system from the solar collector, which is given as:

$$E_u = Q_u \left(1 - \frac{T_0}{T_{sc}} \right) \quad (52)$$

E_E is the exergy associated with the refrigeration output, which is calculated as the working fluid exergy difference across the evaporator.

$$E_E = m_E \cdot [(h_{E,\text{in}} - h_{E,\text{out}}) - T_0(s_{E,\text{in}} - s_{E,\text{out}})] \quad (53)$$

E_H is the heat exergy output, which is given by:

$$E_H = Q_H \left(1 - \frac{T_0}{T_{hc}} \right) \quad (54)$$

3. Methodology

In the present study, the compound parabolic collector is assumed to be located in Xi'an (34.27N, 108.95E) on June 12, and mounted on a horizontal east–west axis and oriented with its aperture plane sloping at an angle of 45°. The concentration ratio of the collector is 5, the width of its absorber plane is 0.06 m and its length is 2 m. The heat collected at the absorber surface is transferred to water flowing through two tubes having an outer diameter of 0.018 m and an inner diameter of 0.014 m attached to the bottom side. In addition, overall loss coefficient U_{lo} is 10.5 W m^{−2} K^{−1}, heat-transfer coefficient on inside of absorber tube is 230 W m^{−2} K^{−1}, transmissivity of the cover

is 0.89, reflectivity of concentrator is 0.87, and absorptivity of absorber surface is 0.94. The number of CPC arrays is assumed to be 1000, that can be designed according to the requirement load for the different applications.

Water is used as storage liquid under the pressure of 0.5 MPa in the sensible heat storage system. The cylindrical hot water storage tank, 0.8 m in diameter and 1.0 m high, is made from a steel plate ($\rho = 7800 \text{ kg m}^{-3}$, $c_p = 0.46 \text{ kJ kg}^{-1} \text{ K}^{-1}$) 0.006 m thick. And it is insulated all round with glass wool insulation 0.02 m thick.

R123 is selected as the working fluid because it is recognizable as a low pressure refrigerant that is nontoxic, nonflammable and non-corrosive.

For the CCHP system simulation, the following assumptions are also made:

- (1) The system reaches a steady state, and pressure drop in pipes and heat losses to the environment in the boiler, turbine, heater, ejector, condenser, regenerator and evaporator are neglected.
- (2) The flow across the throttle valve is isenthalpic.
- (3) The condenser outlet state is saturated liquid, and its temperature is assumed to be approximately 5 °C higher than the environment temperature.
- (4) The working fluid at the evaporator outlet is saturated vapor.

The main assumptions for the CCHP system simulation are summarized in Table 1.

The efficiencies of the ejector are assumed to be $\eta_n = 0.90$, $\eta_m = 0.85$, $\eta_d = 0.85$ by making reference to literatures [8,17,19].

In order to improve the performance of the CCHP system, it is necessary to conduct a parametric optimization to reach the best performance. In the present study, the exergy efficiency, which can evaluate the performance of the CCHP system, is selected as objective function for parameter optimization.

Parameter optimization is achieved by means of genetic algorithm to reach the maximum exergy efficiency. The GA, which is presented firstly by professor Holland [20], is a stochastic global search method that simulates natural biological evolution. Based on the Darwinian survival-of-fittest principle, the genetic algorithm operates on a population of potential solutions to produce better and better approximations to the optimal solution. The GA differs from more traditional optimization techniques because it involves a search from a population of solutions and not from a single point.

The GA encodes a potential solution to a specific domain problem on a simple chromosome-like data structure (which constitutes an individual), where genes are parameters of the problem to be solved. In the present study, the float-point coding is used in parameter optimization for the CCHP system. Each chromosome vector is coded as a vector of floating point numbers of the same length as the dimension of the search space. Chromosome is defined as a real number vector, $X = (x_1, x_2, \dots, x_n)$, $x_i \in R$, $i = 1, 2, \dots, n$, where i is i -th parameter for the CCHP system, n is the number of optimizing parameters.

Table 1

Main assumptions for the CCHP system.

Environment temperature (°C)	20
Environment pressure (MPa)	0.10135
Auxiliary heater outlet temperature (°C)	140
Heater outlet temperature (°C)	70
Refrigeration temperature (°C)	−5.0
Turbine isentropic efficiency (%)	85
Pump isentropic efficiency (%)	70
Pinch point temperature difference (°C)	10.0
Approach point temperature difference (°C)	5.0

The GA uses fitness function to evaluate adaptability of individual without external information in the evolution search. The adaptability is expressed by the fitness value. A bigger fitness value means a better adaptability subjected to constraints and a better viability of the individual. Fitness function which is not constrained by definition domain, continuity and differentiability, requires that the objective function is defined as the form of non-negative maximum. In this optimization, the exergy efficiency is selected as the fitness function.

The GA operators include selection operator, crossover operator and mutation operator. Selection operator is responsible for selecting the parents to create the next generation of solutions. The parent is chosen with a probability based on its fitness. The higher the fitness, the higher is the probability of selection. The rank-based model is selected for this optimization in the CCHP system.

Crossover operator is the basic operator for producing new chromosomes. It produces new individuals that have some parts of both parent's genetic material. The simple arithmetic crossover is applied to this optimization problem due to very simple operation, which is presented as follows:

$$\begin{cases} c_1 = af_1 + (1-a)f_2 \\ c_2 = af_2 + (1-a)f_1 \end{cases} \quad (55)$$

where a is a random number between 0 and 1, f_1 and f_2 are parents individuals which are selected to crossover each other, c_1 and c_2 are children individuals which are produced by crossover.

Mutation is needed because even if selection and crossover together search new solutions, they tend to cause rapid convergence and there is the danger of losing potentially useful genetic material. The role of mutation in GA has been that of restoring lost or unexplored genetic material into the population to prevent the premature convergence of the GA to suboptimal solutions. Random mutation is adopted to optimize the parameters for the CCHP system. It is achieved by selecting individuals from the range of the parameter according to mutation probability.

The steps of GA are made as follows:

- (1) Initialize the population size, crossover probability, mutation probability, stop generation, and generate the initial population randomly.
- (2) Calculate the fitness of each individual for parent generation, and order the fitness.
- (3) Select the individuals from parent generation, and create the children generation using crossover and mutation operators.

Table 2
Results of simulation for the CCHP system.

State	t (°C)	p (MPa)	Dryness	h (kJ kg ⁻¹)	s (kJ kg ⁻¹ K ⁻¹)	m (kg s ⁻¹)
1	54.48	1.0000	0.000	255.981	1.184	0.455
2	130.00	1.0000	1.000	462.514	1.734	0.455
3	94.15	0.3000	1.000	443.589	1.743	0.227
4	78.52	0.0914	1.000	435.018	1.781	0.159
5	25.00	0.0914	0.000	225.139	1.088	0.410
6	25.00	0.0914	0.000	225.139	1.088	0.386
7	25.00	0.0914	0.000	225.139	1.088	0.024
8	-5.00	0.0258	0.164	225.139	1.094	0.024
9	-5.00	0.0258	1.000	378.438	1.666	0.024
10	70.00	0.3000	1.000	424.799	1.690	0.068
11	25.13	0.3000	0.000	225.342	1.088	0.386
12	53.96	0.3000	0.000	255.261	1.184	0.455
13	65.52	0.0914	1.000	425.422	1.754	0.227
14	71.35	0.0914	1.000	429.705	1.766	0.410
15	142.64	0.5000	0.000	600.645	1.767	10.000
16	140.00	0.5000	0.000	589.334	1.739	10.000
17	140.00	0.5000	0.000	589.334	1.739	0.846
18	113.57	0.5000	0.000	478.332	1.458	0.846

Table 3
The performance of the CCHP system.

Turbine work (kW)	12.730
Pump 1 work (kW)	0.079
Pump 2 work (kW)	0.327
Net power output (kW)	12.324
Cooling output (kW)	3.694
Heating output (kW)	1.281
Useful heat gain from solar collector (kW)	110.708
Extraction mass flow ratio	0.5
Mass flow branching ratio	0.7
Heating exergy (kW)	0.224
Cooling exergy (kW)	0.344
Exergy input from solar collector (kW)	32.405
Thermal efficiency (%)	15.63
Exergy efficiency (%)	39.78

- (4) Calculate the fitness of each individual for children generation. If the maximum fitness of children generation is less than that of parent generation, substitute the maximum fitness of parent generation for that of children generation.
- (5) If generation reaches the stop generation, stop the optimization. Otherwise go back to step 3.

4. Results

The simulation of the CCHP system driven by solar energy was carried out using a simulation program written by authors with Fortran. Iterative relative convergence error tolerance was 0.01%. The thermodynamics properties of the working fluid were calculated by REFPROP 6.01 [21] developed by the National Institute of Standards and Technology of the United States.

Table 2 shows the thermodynamic state of the CCHP system driven by solar energy, and Table 3 shows the results of thermodynamic simulation.

Fig. 4 shows the variations of the turbine power, heating output, cooling output and exergy efficiency with the slope angle of the aperture plane for the solar collectors at 10 a.m. ($\omega = -30^\circ$) on June 12 while the other parameters are kept constant as those in Table 1. It can be seen that the turbine power, heating output, cooling output and exergy efficiency increase first to maximum and then decrease as the slope angle of the aperture plane for the solar collectors

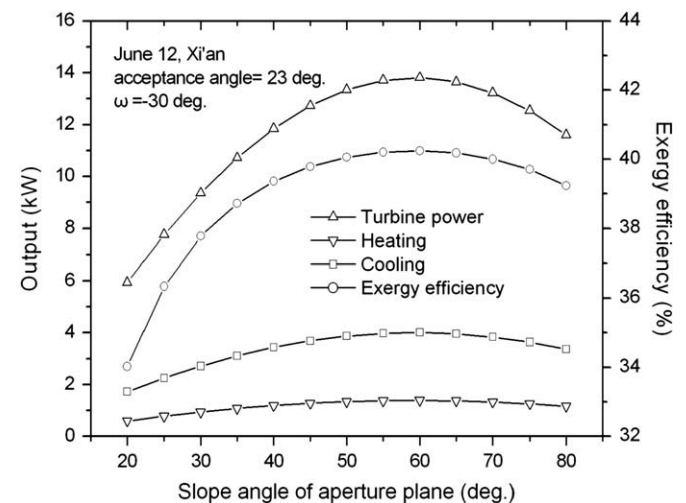


Fig. 4. Effect of the slope angle for the aperture plane on the turbine power, heating output, cooling output and exergy efficiency.

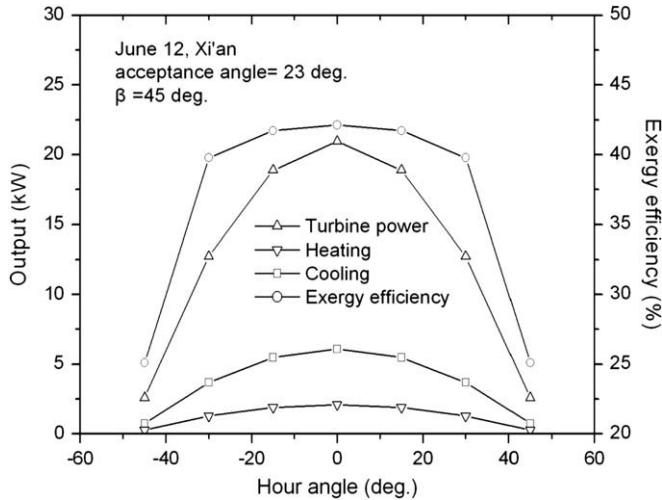


Fig. 5. Effect of the hour angle on the turbine power, heating output, cooling output and exergy efficiency.

increases. The exergy efficiency reach the maximum when the slope angle of the aperture plane is 60° , because the solar collectors can absorb the solar radiation effectively at $\beta = 60^\circ$.

Fig. 5 shows the variations of the turbine power, heating output, cooling output and exergy efficiency with the hour angle at $\beta = 45^\circ$ on June 12 while the other parameters are kept constant as those in Table 1. It is obvious that the turbine power, heating output, cooling output and exergy efficiency increase first to maximum and then decrease as the hour increases. The maximum exergy efficiency takes place at $\omega = 0^\circ$, namely at midday. It is easy to understand that the solar radiation is the strongest at midday.

For the system optimization, the hour angle and the slope angle of the aperture plane for the solar collectors are kept to be the optimal value of 0° and 60° , and the other parameters of the solar collector subsystem are assumed to be constant as those in Table 1. Thus, the parameters chosen for optimizing this CCHP system are turbine inlet pressure, turbine inlet temperature, extraction pressure, extraction mass flow ratio and mass flow branching ratio which could affect the performance of the CCHP system markedly. Table 4 shows the condition of the parameter optimization. Table 5 lists the optimization results for the CCHP system.

In addition, it is shown that the major output of the system is different at any given exergy efficiency. Table 3 shows that at an exergy efficiency of 39.78%, 71.24% of the output is net power. Table

Table 4
The condition of the parameter optimization.

Environment temperature ($^\circ\text{C}$)	20
Environment pressure (MPa)	0.10135
Slope angle for the aperture plane, ($^\circ$)	60
Hour angle, ($^\circ$)	0
Auxiliary heater outlet temperature ($^\circ\text{C}$)	140
Heater outlet temperature ($^\circ\text{C}$)	70
Refrigeration temperature ($^\circ\text{C}$)	-5.0
Turbine isentropic efficiency (%)	85
Pump isentropic efficiency (%)	70
Pinch point temperature difference ($^\circ\text{C}$)	10.0
Approach point temperature difference ($^\circ\text{C}$)	5.0
Population size	50
Crossover probability	0.95
Mutation probability	0.05
Stop generation	200

Table 5

The optimization results of the parameters for the CCHP system.

Turbine inlet pressure (MPa)	1.183
Turbine inlet temperature ($^\circ\text{C}$)	129.62
Extraction pressure (MPa)	0.395
Extraction mass flow ratio	0.735
Mass flow branching ratio	0.231
Turbine work (kW)	18.136
Pump 1 work (kW)	0.104
Pump 2 work (kW)	0.650
Net power output (kW)	17.382
Cooling output (kW)	4.437
Heating output (kW)	77.510
Heating exergy (kW)	13.588
Cooling exergy (kW)	0.414
Exergy input from solar collector (kW)	52.021
Exergy efficiency (%)	60.33

5 shows that at the maximum exergy efficiency of 60.33%, 78.03% of the output is heating.

5. Discussion

The proposed CCHP system is designed to utilize the solar energy to perform the trigeneration based on the crisis of world energy in the future. It can supply the electricity, refrigeration and heating output simultaneously in terms of different energy requirements for an institution (company, plant, hospital, hotel, or business center). The amount of the cooling and heating output could be regulated by controlling the extraction mass flow ratio and mass flow branching ratio to meet the cooling and heating requirements of the users in the summer or winter, and the surplus electricity can be sold to power grid. In the present study, the thermodynamic simulation is conducted to show that the proposed CCHP system is feasible theoretically. However, owing to lack of experimental data for this CCHP system, the experimental study will be carried out to validate the feasibility in the future. In addition, because of using large numbers of compound parabolic collectors to absorb the solar radiation, the investment of the CCHP system may be considerable, but this cannot deny the value of the CCHP system to solve the energy problem. The economic analysis will be conducted to evaluate the feasibility commercially for the CCHP system in the coming research.

6. Conclusion

In the present study, a new combined cooling, heating and power system is proposed. This system is driven by solar energy to reduce the consumption of fossil fuel. The system combines Rankine cycle and ejector refrigeration cycle, which could produce cooling output, heating output and power output simultaneously. The effects of hour angle and the slope angle of the aperture plane for the solar collectors on the system performance are examined. And the parameter optimization is achieved using exergy efficiency as the objective function by means of genetic algorithm.

It is shown that the optimal slope angle of the aperture plane for the solar collectors is 60° at 10 a.m. on June 12, and the CCHP system can reach the optimal performance with the slope angle of 45° for the aperture plane at midday. It is found that the CCHP system has a maximum exergy efficiency of 60.33% under the condition of the optimal slope angle and hour angle, when turbine inlet pressure, turbine inlet temperature, extraction pressure, extraction mass flow ratio and mass flow branching ratio are 1.183 MPa, 129.62 $^\circ\text{C}$, 0.395 MPa, 0.735 and 0.231, respectively.

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